

## MACHINE BUILDING AND MACHINE SCIENCE



UDC 62-19


Original article

<https://doi.org/10.23947/2687-1653-2022-22-2-130-141>

## Determination of the Dynamic Characteristics of a Gear Pump by the Load Variation Method Using Special Bench Systems

Valeriy I. Sanchugov , Pavel D. Rekadze  

Samara National Research University, 34, Moskovskoye Shosse, Samara, Russian Federation

 [Rekadze1993@gmail.com](mailto:Rekadze1993@gmail.com)

### Abstract

**Introduction.** Some of the major factors in the occurrence of vibration of units, causing fatigue failure of the housings of elements, pipelines, and failure of pump elements, are pulsations of the working medium in the hydraulic systems of machine tools, fuel feed systems of aircraft engines and liquid-propellant engine supply. This study aimed at the implementation of a method for determining the dynamic characteristics of a volumetric pump using special bench systems, and the comparison of the calculation results to the experimental data. The stages of calculating the dynamic characteristics of a volumetric pump were described, taking into account the pre-developed special bench systems on the example of an external gear pump with a capacity of 14 cm<sup>3</sup>/rev. The implementation of V. P. Shorin's load variation method using special bench systems developed by the authors with predetermined dynamic characteristics was shown. The main stages of the methodology for determining the dynamic characteristics of a gear pump were described.

**Materials and Methods.** Methods of spectral analysis of pulsating pressure were applied in the work. Pulsations of fluid flow at the pump outlet were determined using the impedance method, the method of load variation, and special bench systems.

**Results.** The paper implemented a technique for determining the dynamic characteristics of a gear pump in the drive shaft speed range of 500–2500 rpm for four harmonic components of the vibration spectrum in a wide range of dynamic loads (from inertial to capacitive nature). The bench systems yielding the calculation of the dynamic characteristics of the pump with a minimum error based on the condition of matching the dynamic load and the source of vibrations were analyzed. The developed approach to the evaluation of the dynamic characteristics of the pump was verified through comparing the calculated and experimental data of pressure pulsations in the bench systems with choke, cavity and an extended pipeline at the pump outlet.

**Discussion and Conclusions.** The method for determining the dynamic characteristics of a volumetric pump was implemented using special bench systems developed by the authors. The research results show that the gear pump under study can be considered as an independent source of flow fluctuations, for which the deviation of its own dynamic characteristics from the average values does not exceed 10% for the first harmonic component.

**Keywords:** volumetric pump, load variation method, bench systems, source of flow fluctuations, pump dynamic characteristics, independent source of fluctuations, flow pulsations, impedance.

**Acknowledgements:** the authors would like to thank the reviewers for the comments, which contributed to quality of the article — the research of the second author was carried out on theme no. FSSS-2020-0015 within the frame of the government task of the Ministry of Education and Science of the Russian Federation.

**For citation:** V. I. Sanchugov, P. D. Rekadze. Determination of the dynamic characteristics of a gear pump by the load variation method using special bench systems. Advanced Engineering Research, 2022, vol. 22, no. 2, pp. 130–141. <https://doi.org/10.23947/2687-1653-2022-22-2-130-141>

**Introduction.** The main disadvantages of pumps are the acoustic noise that accompanies their operation, the vibration of the body, connected pipelines, and the resulting structural damage.

The traditional way of improving pump designs, used in mechanical engineering and consisting in smoothing out the uneven flow of the liquid, is hampered by the lack of means of direct measurement of fluid flow at the pump outlet. This limits any experimental evaluation of the design efficiency and finishing work on pumps. In addition, this limits the analytical description of the pump as a source of vibrations required by hydraulic system developers to provide the target dynamic quality of the systems by themselves.

For this reason, the research on the modeling of dynamic processes in hydromechanical systems [1, 2] and the application of new approaches to solving problems of hydraulic system dynamics [3, 4] based on physical principles are pressing issues.

**Materials and Methods.** The implementation of V. P. Shorin's method of load variation [4] is presented. It consists in determining pressure pulsations behind a pump interacting with several bench systems, whose dynamic characteristics are calculated in advance. Further, the processed frequency-response functions of the pressure using the impedance of the bench systems are recalculated into the pump's own dynamic characteristics — pulsation performance and impedance. The development of the approach provides that special bench systems are proposed. They take into account the connecting fittings, adapters and internal channels of the pump, which makes it possible to more accurately determine the dynamic characteristics of pumps, as well as to verify the applicability of the pump model used in the form of an equivalent source of vibrations.

### Research Results

#### *Method for determining the dynamic characteristics of a gear pump*

Samara University has developed a computational and experimental method for determining the dynamic characteristics of a gear pump (pulsation performance and impedance), which is based on the research of L. Thévenin, E. Norton and V. P. Shorin [5, 6]. The developed methodology includes the following main stages: creation of models and calculation of dynamic characteristics of special bench systems; experimental determination of pressure pulsations at the pump outlet; calculation of the spectrum of excited vibrations in various bench systems; determination of approximating dependences for individual harmonic components; calculation of the pump own dynamic characteristics (pulsation performance and impedance); evaluation of the independence (stability) of the dynamic characteristics of the pump from the characteristics of bench systems; verification of the applicability of the pump model used as an equivalent source of flow (or pressure) fluctuations. The following is an example of the implementation of the proposed technique using an external gear pump with a specific capacity of 14 cm<sup>3</sup>/rev.

#### *Models and calculation of dynamic characteristics of special bench systems*

Basic options of bench systems at the liquid outflow during pump tests were considered. In concentrated parameters, the bench system can implement active flow-dependent, inertial, and elastic loads. In distributed parameters, the bench system can be provided in the form of an extended cylindrical pipeline implementing an active frequency-independent non-reflective load.

Pump models in the form of equivalent source of flow fluctuations (ESFF) or pressure (ESPF) are determined using constant flow source  $Q_0$ , an ideal source of pressure fluctuations  $p$  (or flow  $q$ ) and internal impedance  $Z_u$  [4]. Figure 1 shows pump models in the form of an equivalent source of pressure fluctuations (and flow) with an attached system.

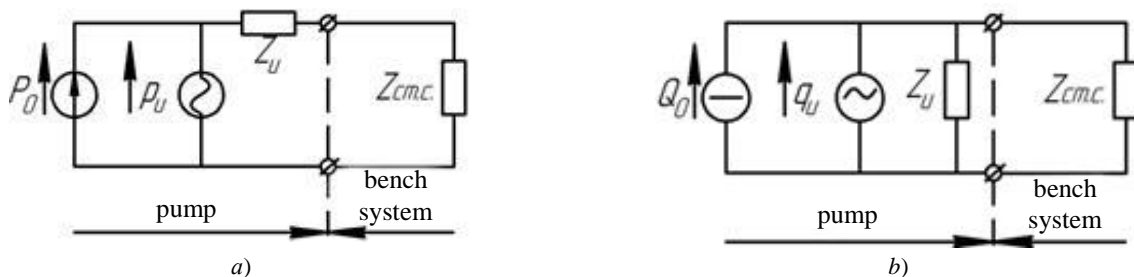


Fig. 1. Pump models in the form of an equivalent source of: *a* — pressure fluctuations; *b* — flow with attached system

$Q_0, P_0$  — average flow and pressure;  $p_H, q_H$  — pump's own pulsating performance in terms of pressure and flow,

$Z_u, Z_{cm.c.}$  — impedance of pump and bench system (adapted from [4])

In this connection, the task of selecting a bench system is to provide such system parameters at which the best conditions for high measurement accuracy are realized. The task of selecting the parameters of the attached system should be carried out when the impedance changes in the range from  $-\infty$  to  $+\infty$ .

Figure 2 shows schematic diagrams of the bench systems.

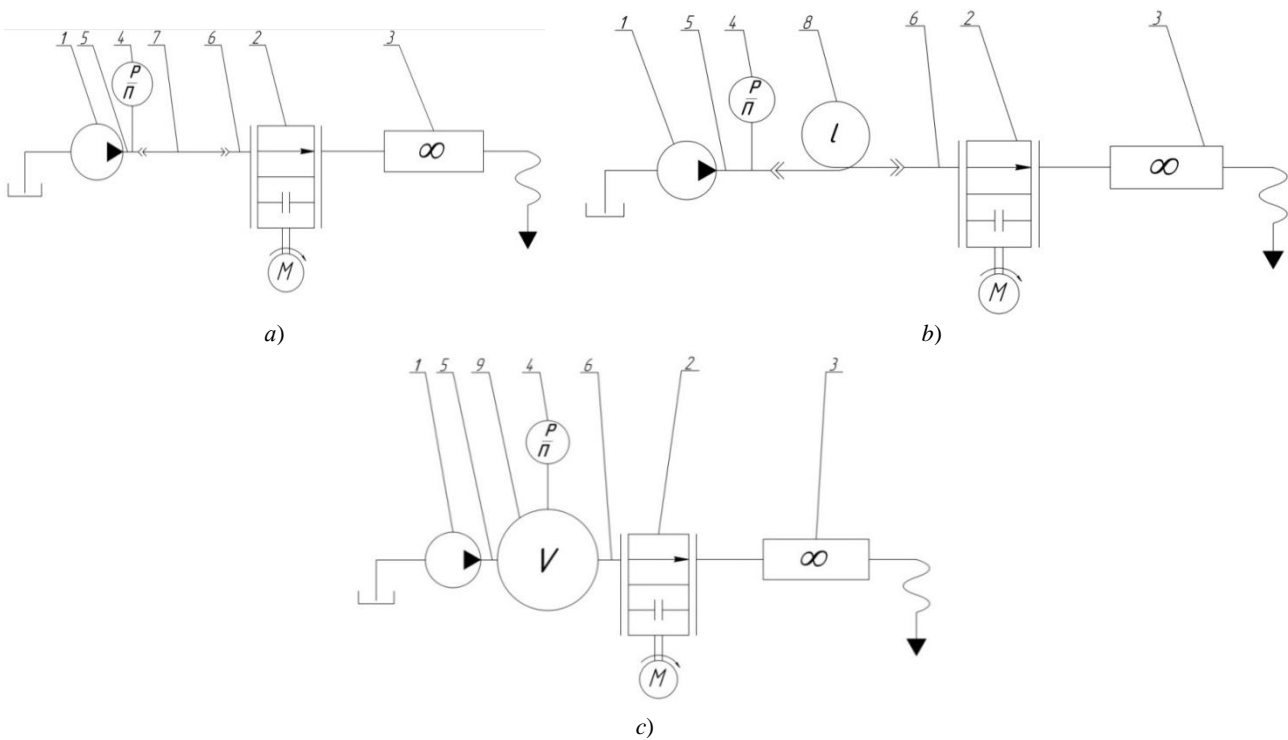


Fig. 2. Schematic diagrams of the bench systems for determining own dynamic characteristics of pumps: *a* — with active throttle load; *b* — with active non-reflective load; *c* — with capacitive load: 1 — test pump; 2 — load choke; 3 — flow meter; 4 — pressure sensor; 5, 6 — connecting fittings; 7 — main line; 8 — extended pipeline; 9 — cavity

Computational models of impedances of special bench systems:

– with active throttle load (throttle):

$$Z_{cm.c.k}(Q_0, f) = \sqrt{(Z_{op}(Q_0) + ReZ_{mp}(Q_0))^2 + \left(2\pi f_k \frac{\rho l_{\Sigma}}{S_{mp}}\right)^2} e^{j \arctg \frac{2\pi f_k \frac{\rho l_{\Sigma}}{S_{mp}}}{Z_{op}(Q_0) + ReZ_{mp}(Q_0)}} \quad (1)$$

– with active distributed load (extended pipeline):

$$Z_{cm.c.} = \frac{\rho a}{S_{mp}}, \quad (2)$$

– with capacitive load (cavity):

$$Z_{cm.c.k}(Q_0, f_k) = \sqrt{\left(ReZ_{lam}(Q_0) + ReZ_{mypo}(Q_0) + ReZ_{m.c.}(Q_0) + Z_{op}(Q_0)\right)^2 + \left(\frac{(2\pi f_k)^2 LC - 1}{2\pi f_k C}\right)^2} * e^{j \arctg \left[ \frac{(2\pi f_k)^2 LC - 1}{2\pi f_k C (ReZ_{lam} + ReZ_{mypo}(f_k) + ReZ_{m.c.}(f_k) + Z_{op}(Q_0))} \right]} \quad (3)$$

The formulas indicate:  $Z_{op}(Q_0)$  — impedance of the throttle;  $ReZ_{Tp}(Q_0)$  — real part of the impedance of the pipeline, including the connecting line, fittings, internal channels in the pump and throttle;  $ReZ_{lam}(Q_0)$ ,  $ReZ_{mypo}(Q_0)$ ,  $ReZ_{m.c.}(Q_0)$  — real part of the impedance of the fittings and the internal channel in the pump and cavity with laminar, turbulent fluid flow behind the pump and local resistances, respectively;  $f_k$  — frequency  $k$  — *th* of the harmonic component of the vibration spectrum behind the pump;  $\rho$  — density of the working environment;  $l_{\Sigma}$  — pipeline length, including the length of main line 6, connecting fittings 5 and 7 and internal channels of the pump and throttle (not shown in Fig. 7);  $S_{mp}$  — cross-sectional area of the pipeline;  $L$  — “inertia” of

the connecting fittings and the internal channel in the pump and cavity;  $C$  — elasticity of the cavity;  $k$  — number of the harmonic component of the vibration spectrum;  $j$  — imaginary unit ( $j = \sqrt{-1}$ ).

Using a bench system with a throttle at the pump outlet, three inertially active dynamic loads were implemented, in which the active resistance was changed through changing the overlap area of throttle  $F_0$ :

- for bench system no. 1, overlap area  $F_{01}=9.290 \times 10^{-6} \text{ m}^2$ ,
- for bench system no. 2, overlap area  $F_{02}=7.299 \times 10^{-6} \text{ m}^2$ ,
- for bench system no. 3, overlap area  $F_{03}=2.247 \times 10^{-5} \text{ m}^2$ .

Using a bench system with a cavity at the outlet of the pump with a volume of 2 liters, an inertial elastic dynamic load was realized.

An active frequency-independent load was implemented using a bench system with an extended pipeline length of 106 m.

The dependences of the impedance (module and phase) on the frequency were calculated according to the models of bench systems using formulas (1)–(3). Table 1 for the 1st harmonic shows the regression functions of the impedance module of the bench system  $Z_i(f)$  and phases  $\beta_i(f)$  for the analyzed bench systems.

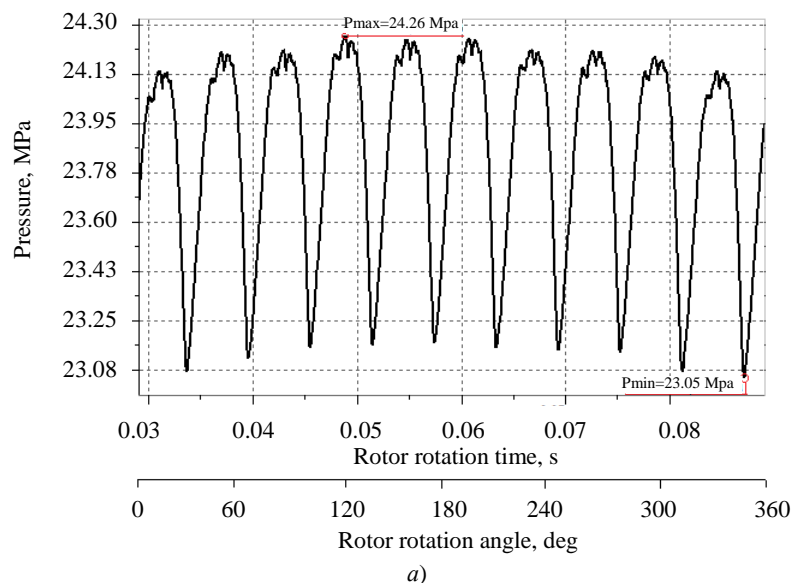
Table 1

Regression functions of dynamic characteristics of bench systems according to the 1st harmonic

No. of stand system according to Figure 2	Bench system type	Bench system impedance module $Z_i(f), \frac{\text{kg}}{\text{m}^4 \text{s}^2}$	Bench system impedance phase $\beta_i(f)$ , rad
a – no.1	With throttle at pump outlet	$1.412 \cdot 10^7 f + 2.201 \cdot 10^9$	$7.066 \cdot 10^{-4} f + 0.199$
a – no.2		$1.610 \cdot 10^7 f + 4.258 \cdot 10^9$	$6.906 \cdot 10^{-4} f + 0.09039$
a – no.3		$4.980 \cdot 10^7 f + 1.503 \cdot 10^9$	$2.723 \cdot 10^{-3} f + 0.251$
b – no.4	With cavity at pump outlet	$3.308 \cdot 10^4 f^2 - 1.416 \cdot 10^7 f + 1.873 \cdot 10^9$	$\arctg(\frac{4.234 \cdot 10^6 f + 1.495 \cdot 10^9}{1.430 \cdot 10^5 f + 2.930 \cdot 10^6})$
c – no.5	With extended pipeline at pump outlet	$1.897 \cdot 10^{10}$	0

#### Experimental determination of pressure pulsation at the pump outlet

The background information for determining the dynamic characteristics of the pump was the measurement of pressure at the pump outlet. An example of an oscillogram and a pressure pulsation spectrum at the pump outlet in the bench system with an active distributed load is shown in Figure 3. Operating mode of the pump in this case is as follows: average pressure  $P_0 = 23.8 \text{ MPa}$ , drive shaft speed  $n = 1000 \text{ rpm}$ .



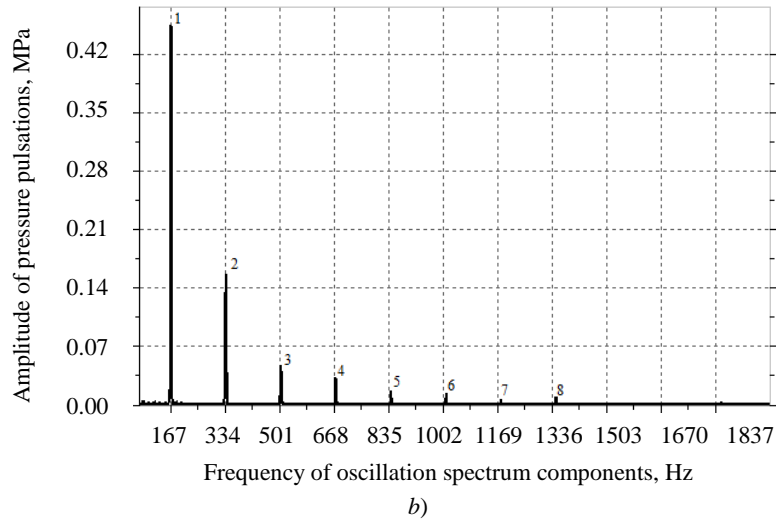


Fig. 3. Pressure at the pump outlet: *a* — oscillogram; *b* — spectrogram

The amplitude-frequency response of pressure pulsations at the pump outlet in the speed range of the pump drive shaft  $n=500\text{--}1250$  rpm is shown in Figure 4. The regression functions built in the *Microsoft Excel* environment using the Trendline tool are plotted with a solid line.

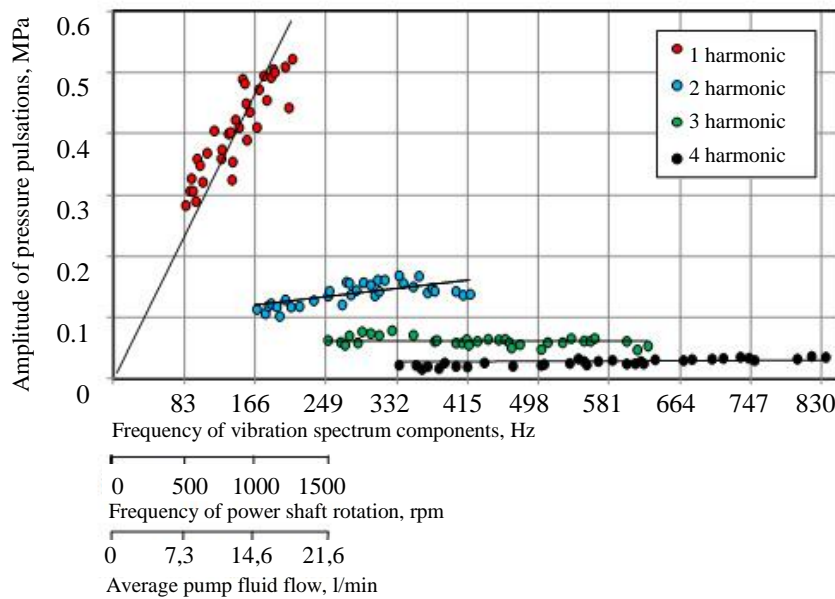


Fig. 4. Amplitude-frequency response of pressure pulsations at the pump outlet by individual harmonic components (dots indicate data on: ● — first, ● — second, ● — third, ● — fourth harmonics, lines indicate regression functions)

The analysis of the oscillogram, spectrogram and amplitude-frequency response of pressure pulsations showed the following: the type of process was polyharmonic steady; the number of recorded harmonics was 8, analyzed<sup>1</sup> — 4; the amplitude range of pressure fluctuations was 0.28–0.52 MPa; the drive shaft frequency range was 500–1250 rpm, the analyzed harmonic components of the spectrum 83–833 Hz.

Fourier series expansion was used for harmonic analysis of periodic signals [7]. To decompose into a Fourier series, sections of the pressure oscillogram measured relative to the reference signal at the rotor frequency were used, i.e., one section of the oscillogram was extracted for a full rotation of the rotor. The reference signal was recorded at a fixed gear position, set at the beginning of recording the pressure waveform for five bench systems. In this way, the amplitude-frequency (FRF)  $A_k(\omega)$  and phase-frequency (PFC)  $\phi_k(\omega)$  characteristics were determined.

<sup>1</sup> It is assumed in the work that the extreme highest analyzed harmonic in amplitude should not be more than 5 % of the amplitude of the first harmonic.

Calculation of the spectrum of excited vibrations in various bench systems and determination of approximating for individual harmonic components of pressure fluctuations

Based on the experimental determination of pressure in five bench systems: with a throttle (no.1–3), with a cavity (no.4) and an extended pipeline (no.5) at the pump outlet, the spectra of excited vibrations were calculated. Table 2 shows the regression functions of the amplitudes  $A_p(f)$  and the initial phases of pressure pulsations for the 1st harmonic.

Table 2

Regression functions of amplitudes and initial phases of pressure fluctuations

No. of stand system according to Figure 2	Bench system type	Pressure pulsation amplitude $A_{pi}(f), \frac{\text{kg}}{\text{ms}^2}$	Initial phase of pressure pulsations $\varphi_i(f), \text{rad}$	Frequency range
a – no.1	With throttle at pump outlet	$1.003 \cdot 10^{-3}f$	–0.543	83–417
a – no.2		$1.243 \cdot 10^{-3}f$	–0.280	
a – no.3		$6,129 \cdot 10^{-4}f$	–5.941	
b – no.4	With cavity at pump outlet	$1.343 \cdot 10^{-6}f^2 - 4.298 \cdot 10^{-4}f + 3.913 \cdot 10^{-2}$	–5.373	
c – no.5	With extended pipeline at pump outlet	$2.694 \cdot 10^{-3}f$	–0.658	

Calculation of the dynamic characteristics of the pump

To calculate the dynamic characteristics of the pump (impedance and pulsation performance of the source), it is required to conduct an experiment with at least two dynamic loads<sup>2</sup>. Therein, according to the model of an equivalent source of flow fluctuations refined by the authors using analytical dependences  $p_i(f)$  and  $Z_i(f)$  from Tables 1 and 2, the pump impedance is calculated from the formula:

$$Z_u = Z_1 \frac{1 - \frac{p_1}{p_2} \frac{Z_2}{Z_1}}{\frac{p_1}{p_2} - \frac{Z_1}{Z_2}}, \quad (4)$$

where  $Z_u$  — pump impedance; ( $Z_u = |Z_u| e^{j\varphi_u}$ ) — pump impedance module; ( $|Z_u| = \sqrt{\text{Re}Z_u^2 + \text{Im}Z_u^2}$ ),  $\text{Re}Z_u, \text{Im}Z_u$  — real and imaginary part of pump impedance;  $\varphi_u$  — pump impedance argument;  $Z_1, Z_2$  — input impedance of dynamic loads no. 1–2; ( $Z_1 = |Z_1| e^{j\beta_1}, Z_2 = |Z_2| e^{j\beta_2}$ ),  $\beta_1, \beta_2$  — impedance arguments of dynamic loads no. 1–2;  $p_1, p_2$  — pressure pulsations behind the pump in bench system no. 1–2; ( $p_1 = A_{p1} e^{j\varphi_1}, p_2 = A_{p2} e^{j\varphi_2}$ ),  $A_{p1}, A_{p2}$  — amplitudes of flow pulsations behind the pump in bench system no. 1–2;  $\varphi_1, \varphi_2$  — initial phases of pressure pulsations of dynamic loads no. 1–2.

Then, using the analytical dependences from Tables 1 and 2, the pulsation capacity of the pump was determined according to the model of an equivalent source of vibrations refined by the authors (variable component of volumetric flow  $q_u$  and pressure  $p_u$ ):

$$q_u = \frac{p_2}{Z_2} \frac{1 - \frac{Z_2}{Z_1}}{1 - \frac{p_2}{p_1}}, \quad p_u = \frac{1 - \frac{Z_1}{Z_2}}{\frac{p_1}{p_2} - \frac{Z_1}{Z_2}}, \quad (5)$$

<sup>2</sup> For a sufficient sample size, it was required to use at least four dynamic loads  $Z_{ij}$ , that form six non-repeating (including  $i \neq j$ ) combinations:  $Z_{12}, Z_{13}, Z_{14}, Z_{23}, Z_{24}, Z_{34}$ .

where  $q_u, p_u$  — variable component of the volume flow ( $q_u = A_{qu} e^{j\varphi_q}$ ) and pressure ( $p_u = A_{pu} e^{j\varphi_p}$ ) of the pump;  $A_{qu}, A_{pu}$  — pump flow and pressure pulsation amplitude;  $\varphi_q, \varphi_p$  — argument of the variable component of the volume flow and pressure of the pump.

Of the calculated characteristics according to ESFF ( $q_u(f)$ ) and ESPF ( $p_u(f)$ ) models, using formula (5), only one will be stable, since the equivalent oscillation source has an independent dynamic characteristic only for one of the parameters (pulsating flow or pressure).

The calculation of impedance  $Z_u$  and the variable component of the volume flow  $q_u$  of the pump was carried out according to formulas (4)–(5) for five dynamic loads, i.e., using ten different combinations of these loads: “1\_2”, “1\_3”, “2\_3”, “1\_4”, “2\_4”, “3\_4”, “1\_5”, “2\_5”, “3\_5”, “4\_5” (Table 1).

For further calculation and analysis, a characteristic of the pump flow pulsation amplitudes  $A_{qu}$  was obtained from ten flow pulsation amplitude curves. The values of the amplitudes  $A_{qu}$  (at each frequency) were calculated using the formula from source [8]:

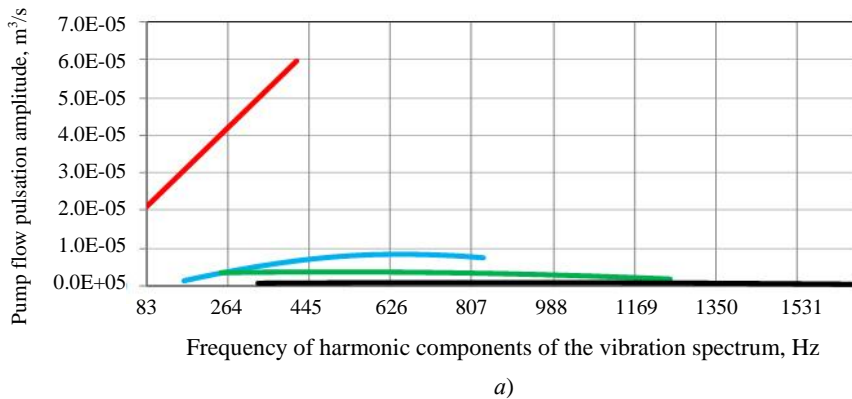
$$A_{qu} = \overline{A_{qu}} \pm \Delta A_{qu},$$

where  $\overline{A_{qu}}$  — average amplitude of pulsating flow;  $\Delta A_{qu}$  — confidence interval of the amplitude of pump flow pulsations ( $\Delta A_{qu} = \frac{t\sigma}{\sqrt{m}}$ );  $\sigma$  — standard deviation of the amplitude of the pulsating flow rate from the average value;  $m$  — number of dynamic load combinations ( $m=10$ );  $t$  — Student's coefficient ( $t=2.262$ , was calculated for the confidence level 0.95).

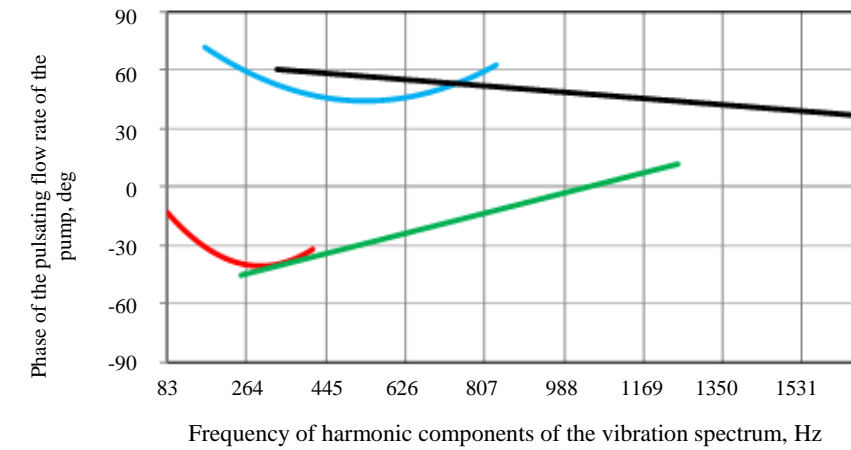
The confidence interval of amplitudes  $\Delta A_{qu}$  was formed from errors (instrument and method) and the degree of dependence of the dynamic characteristics of the pump  $A_q(f)$  on the bench system. The instrument error consisted of the error of determining the pressure and phases, the method error included the error of the unaccounted influence of the dynamic properties of the drive mechanical system.

The phases of flow pulsations  $\varphi_q$  and their deviations were calculated in a similar way. When calculating the phases, combinations “1\_2” and “1\_5” were excluded due to the degeneration of the variable component of the volume flow of the pump  $q_u$  when using close values of characteristics at the same frequency:  $\varphi_1 \approx \varphi_2$  and  $\beta_1 \approx \beta_2$ .

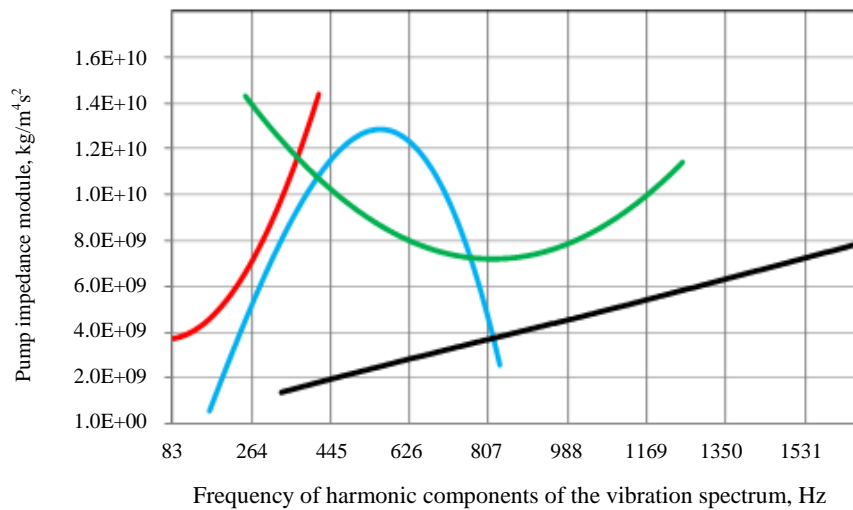
The results of calculating the intrinsic dynamic characteristic of the pump in the form of pulsation performance and impedance are shown in Figure 5. Phases, for convenience, are presented in degrees.



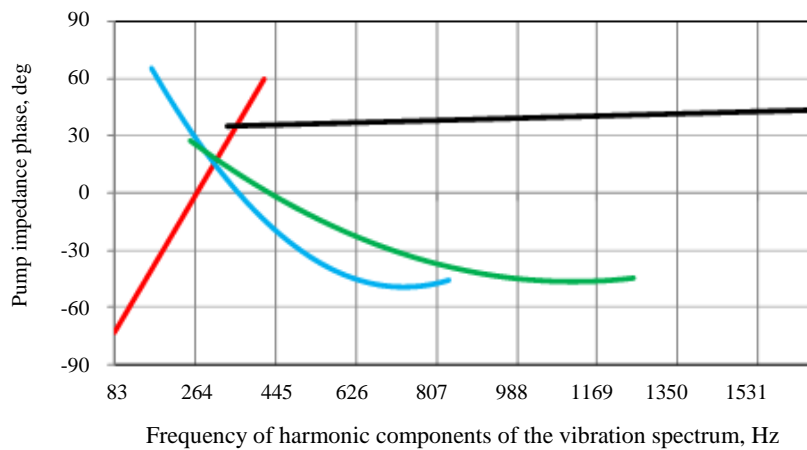




b)



c)



d)

Fig. 5. Dynamic characteristics of the pump: *a* — flow pulsation amplitude; *b* — flow pulsation phase; *c* — impedance modulus; *d* — impedance phase (solid lines indicate data on: — first, — second, — third, — forth harmonics)

The amplitude of the flow pulsations at the 1st harmonic has a monotonically increasing character. With an increase in the harmonic number, the slope of the flow pulsation amplitude curve decreases to an almost constant value to the 4th harmonic. The values of amplitudes  $A_q$  at the higher harmonics are by an order of magnitude less than at the 1st harmonic.

The pump impedance module at the 1st harmonic has a monotonically increasing character.



To evaluate the realized dynamic loads, the criterion of matching the impedance module  $|Z_u|$  of the oscillation source and the input impedance module of the dynamic load  $|Z_{cm.c}|$  was used [99, 10]. The impedance modules should be comparable ( $|Z_u| \cong |Z_{cm.c}|$ ), so that the quantitative values of the dynamic characteristics of the pump were determined with minimal error. For comparison, ratio  $\left| \frac{Z_u}{Z_{cm.c}} \right|$  was used. At  $\left| \frac{Z_u}{Z_{cm.c}} \right| \cong 1$ , the dynamic load was considered consistent with the source.

Due to the difference in the curve shapes of the pump impedance modules and dynamic loads in the frequency range of 83–417 Hz, it was rational to use average ratio  $\left| \frac{Z_u}{Z_{cm.c}} \right|_{f_{min} \dots f_{max}}$ , that provided considering the contribution of most of

the frequency range, i.e., ratio  $\left| \frac{Z_u}{Z_{cm.c}} \right|_{f_{min} \dots f_{max}}$  according to the 1st harmonic would be:

- for active throttle load no.1 — 1.16;
- for active throttle load no.2 — 0.8;
- for active throttle load no.3 — 2.65;
- for capacitive load no.4 — 4.64;
- for active distributed load no.5 — 0.35.

That is, the dynamic loads of bench systems no. 1, 2 were the most consistent with the source of vibrations. Thus, the dynamic characteristic of the pump, calculated using dynamic loads of bench systems no.1, 2 (“1\_3”, “2\_3”, “1\_4”, “2\_4”, “1\_5”, “2\_5”) was determined with a minimum error.

#### *Evaluation of the stability of the dynamic characteristics of the pump from the characteristics of bench systems*

Stability of the dynamic characteristics  $A_{qu}$  and  $\varphi_q$  ( $|Z_u|$  and  $\varphi_u$ ) of the pump was evaluated using mathematical statistics tools [8]. Thus, the stability of the characteristic  $A_{qu}$  was estimated through calculating the ratio of the confidence interval  $|\Delta A_{qu}|$  at frequency  $f$  to the average value of the characteristic  $\overline{A_{quMe}}$  at this frequency<sup>3</sup>. The resulting relative deviation  $\Delta_q$  should not exceed the specified deviation  $\Delta_{dong}$ :

$$\Delta_q = \left| \frac{\Delta A_{qu}}{\overline{A_{quMe}}} \right| \leq \Delta_{dong},$$

where  $\overline{A_{quMe}}$  — average median value of the amplitude of pump flow pulsations;  $\Delta_{dong}$  — specified relative deviation of the pump flow rate pulsation amplitude from the average value.

With relative deviation  $\Delta_q \rightarrow 0$  (when  $|\Delta A_{qu}| \rightarrow 0$ ), the stability of characteristic  $A_{qu}$  was maximum. Based on the experience of other researchers, it is preferable to select value  $\Delta_{dong}$  within 5–30 % [11–17].

For a single-digit evaluation of the dynamic characteristics of the pump, it is rational to use the value of the relative deviation. The calculation of such a value is possible through estimating the average median value of relative deviations

in the frequency range of each harmonic ( $f_{min} \dots f_{max}$ ):  $\Delta_{qMe} = \left| \frac{\Delta A_{qu}}{\overline{A_{qu}}} \right|_{f_{min} \dots f_{max}}$ . Value  $\Delta_{qMe}$  is efficiently used, if

$$\left| \frac{\Delta A_{qu}}{\overline{A_{qu}}} \right|_{f_{min} \dots f_{max}} \leq 0.3, \text{ i.e., when the confidence interval } \Delta A_{qu} \text{ does not exceed 30\% of the average amplitude of pulsating } \overline{A_{qu}}.$$

The results of assessing the stability of the dynamic characteristics of the pump from the characteristics of the bench systems<sup>4</sup> were: according to the 1st harmonic — 2 %, according to the 2nd — 8 %, according to the 3rd — 15 %, according to the 4th — 48 %. The results of the calculation of the dynamic characteristics of the pump showed that for

<sup>3</sup> Stability of other intrinsic characteristics of the pump is evaluated similarly.

<sup>4</sup> It is required to consider the method error of the approach and the instrument error of determining the pressure.

four harmonics in the frequency range of 500–2500 rpm, the gear pump under study should be considered according to the model of an equivalent source of flow fluctuations.

*Checking the applicability of the pump model used as an equivalent source of flow fluctuations*

To test the developed approach for evaluating the dynamic characteristics of the pump, a calculated determination of pressure pulsations in bench systems with a throttle, a cavity and an extended pipeline at the pump outlet was performed. Using the found dynamic characteristics of the pump, the amplitudes of pressure pulsations in the analyzed bench systems were calculated, which were compared to the experimental ones. The results of the calculation of amplitudes  $A_p(f)$  in a bench system with an extended pipeline are presented below. The amplitudes of pressure pulsations  $p_i$  were calculated in accordance with the ESFF model from the formula:

$$p_i = \frac{q_u \cdot Z_u \cdot Z_i}{Z_u + Z_i}, \quad (6)$$

The results of calculating the amplitudes of pressure pulsations behind the pump are shown in Figure 6. The dotted line indicates the amplitudes obtained experimentally, and the solid line — calculated from formula (6).

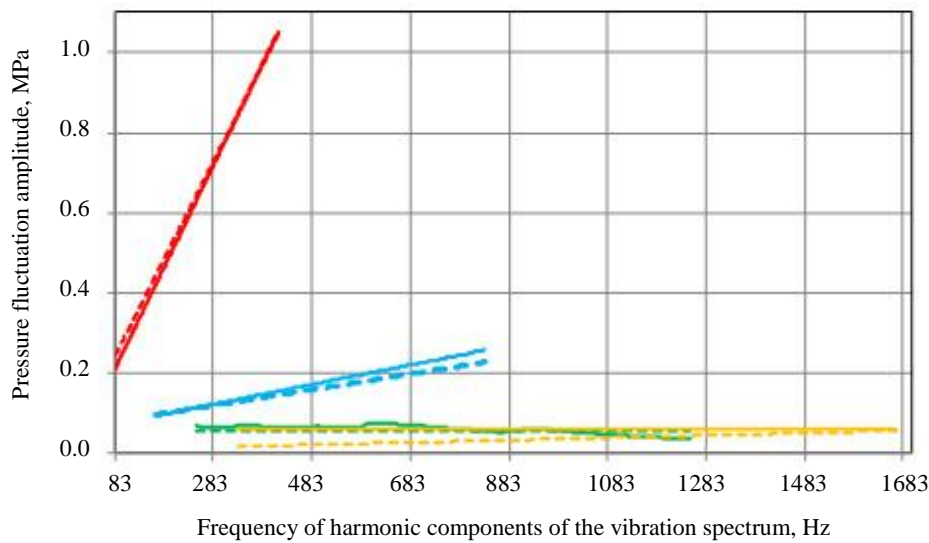


Fig. 6. Pressure frequency response for the bench system with extended pipeline length  $l=106$  m (solid lines indicate calculated data on: — first, — second, — third, — forth harmonics; dotted lines indicate experimental data)

Convergence of the results of the verification calculation was evaluated according to the formula of deviation of the difference between the experimental and calculated values from the experimental data. The results of the test calculation for the bench systems with a throttle, a cavity and an extended pipeline at the pump outlet showed that according to the 1st harmonic, the relative deviation from the experimental data for the considered bench systems did not exceed 10 %, the minimum relative deviation was characteristic of a bench system with an extended pipeline at the pump outlet — 2%. The amplitudes of the pressure pulsations of the 2nd–4th harmonics were within the corridor calculated from the formula  $A_p = \overline{A_p} \pm \overline{A_p}$  (where  $\overline{A_p}$  — the average amplitude of the pulsating pressure).

**Discussion and Conclusions.** Methodology for calculating dynamic characteristics of the pump in the form of an equivalent source of fluid flow fluctuations was implemented on the example of a gear pump with a capacity of 14 cm<sup>3</sup>/rev using special bench systems developed by the authors and an updated model of impedance and pulsation flow rate. It was shown that the investigated gear pump of external gearing was rationally considered according to the model of an equivalent source of flow fluctuations. The proposed approach was verified in the form of calculation of pressure pulsation amplitudes in the bench systems and comparison to experimental values. The deviation of the calculated amplitudes of pressure pulsations from the experimental values did not exceed 10%. It is worth noting that the impact on the dynamic characteristics of the gear pump of the drive mechanical system and the wear of the gear pump parts remained outside the scope of the review. Based on the work carried out, it was possible to formulate directions for the development of the proposed approach:

- assessment of the dynamic characteristics of the gear pump during the entire service life;

- development of a four-pole model of a gear pump as an equivalent source of vibrations, taking into account the mechanical drive and hydraulic output systems;
- development of a gear pump model as an equivalent source of vibrations, taking into account its design features, the displacement process, reverse water hammer in the chamber between teeth, the work of the locked volume, and leakage from the injection area to the suction area.

Here, it is required to expand the nomenclature of the tested volumetric pumps for the entire class of volumetric displacement machines to confirm the operability of the proposed approach.

## References

1. Rybak AT, Zharov VP, Fridrikh RA. Modelirovanie i ehksperimental'nye issledovaniya gidromekhanicheskoi sistemy so znakoperemennoi nagruzkoi. Vestnik of DSTU. 2006;6:17–25. (In Russ.)
2. Rybak AT. Ob"emnaya zhestkost' i ee vliyanie na dinamiku gidromekhanicheskoi sistemy. Vestnik of DSTU. 2006;6:200–207. (In Russ.)
3. Ozerskiy AI. Application of Lagrange approach to solving problems of heat power plants hydraulic systems dynamics. Vestnik of DSTU. 2010;10:914–925.
4. Bystrov ND, Gimadiev AN, Golovin AN, et al. Formirovanie dinamicheskikh svoistv truboprovodnykh tsepei. Shorin VP (ed.) Samara: Samara University Publ.; 2020. 328 p. (In Russ.)
5. Staniloiu M, Popescu H, Rezmeriță G, et al. The equivalent circuits Thévenin and Norton. The Scientific Bulletin of Electrical Engineering Faculty. 2021;21:40–48.
6. Thévenin L. Sur un nouveaux théorème d'électricité dynamique. Comp. Rendus hebdomadaires des Séances de l'Académie des Sci. 1883;97:159–161.
7. Oberguggenberger M, Toft J, Vindas J, et al. (eds.) Generalized Functions and Fourier Analysis. Springer International Publishing AG, 2017. 276 p.
8. Bronshtein IN, Semendyayev KA, Musiol G, et al. Handbook of Mathematics, 5th ed. Berlin: Springer, 2015. 1207 p.
9. Artyukhov AV, Brudkov LI. Osobennosti chastotnykh ispytaniy gidravlicheskh nasosov. In: Interuniversity coll. of sci. papers "Vibratsionnaya prochnost' i nadezhnost' dvigatelei i sistem letatel'nykh apparatov". Kuibyshev: KUI. 1982;9:12–17. (In Russ.)
10. Sundararajan D. Introductory circuit theory. Switzerland: Springer; 2020. 297 p. <https://doi.org/10.1007/978-3-030-31985-4>
11. Ichianagi T, Nishiumi T, Nakagawa Sh. The temperature characteristics of flow ripple and source impedance in an external gear pump. In: Proc. 10th JFPS Int. Symposium on Fluid Power, 2017. 5 p.
12. Artyukhov AV, Shorin VP. Metodika opredeleniya dinamicheskikh kharakteristik gidravlicheskh nasosov. In: Interuniversity coll. of sci. papers "Dinamicheskie protsessy v silovykh i ehnergeticheskikh ustanovkakh letatel'nykh apparatov apparatov". Kuibyshev: KUI; 1988. P. 70–77. (In Russ.)
13. Jinghao Liu. Source flow ripple and source impedance measurement for different hydraulic pumps. In: Proc. 47th Int. Congress and Exposition on Noise Control Engineering (InterNoise 2018). Chicago, Illinois; 2018. 7 p.
14. Liselott Ericson. On Fluid Power Pump and Motor Design – Tools for Noise Reduction. PhD Thesis. Linköping University, Sweden; 2011. 130 p.
15. Bramley C, Johnston DN. Comparison of methods for measuring pump flow ripple and impedance. In: Proc. ASME/BATH 2017 Symposium on Fluid Power and Motion Control. 2017. 11 p. <http://dx.doi.org/10.1115/FPMC2017-4223>
16. Jinghao Liu, Sanghoon Suh, Yuzhen Yang. Hydraulic Fluid-Borne Noise Measurement and Simulation for Off-Highway Equipment. In: Proc. 2017 SAE/NOISE-CON Joint Conf. (NoiseCon 2017), Grand Rapids, MI. 2017;9:435–443.
17. Min Pan, Nigel Johnston, Andrew Plummer, et al. Theoretical and experimental studies of a switched inertance hydraulic system including switching transition dynamics, non-linearity and leakage. Proc. Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering. 2014;228:802–815. <https://doi.org/10.1177/0959651814548299>

Received 27.04.2022

Revised 20.05.2022

Accepted 20.05.2022

*About the Authors:*

**Sanchugov, Valeriy I.**, professor of the Automatic Systems of Power Plant Department, Samara National Research University (Samara University) (34, Moskovskoye Shosse, Samara, 443086, RF), Dr.Sci. (Eng.), [ORCID](#), [Sanchugovv.i@yandex.ru](mailto:Sanchugovv.i@yandex.ru)

**Rekadze, Pavel D.**, design engineer of Research Institute-201, Samara National Research University (Samara University) (34, Moskovskoye Shosse, Samara, 443086, RF), [ResearcherID](#), [ScopusID](#), [ORCID](#), [Rekadze1993@gmail.com](mailto:Rekadze1993@gmail.com)

*Claimed contributorship*

V. I. Sanchugov: academic advising; basic concept formulation; research objectives and tasks. P. D. Rekadze: basic concept formulation; computational analysis; text preparation; formulation of conclusions; analysis of the research results; the text revision; finalization of conclusions.

*Conflict of interest statement*

The authors do not have any conflict of interest.

*All authors have read and approved the final manuscript.*